PERFORMANCE EVALUATION OF A DIESEL ENGINE WITH VARYING SHIMS CONFIGURATION FOR DIFFERENT BLENDS OF DIESEL AND ETHANOL

MAHENDRA SENGAR, SUMAN SOURAV, VAIBHAV KHANTE & VISHAL NAWANDAR

Department of Mechanical Engineering, Shri Ramdeobaba College of Engineering and Management,

Nagpur, Maharashtra, India

ABSTRACT

Environmental concerns and limited amount of petroleum fuels have caused thick interests in the development of renewable fuels as an alternative for internal combustion (IC) engines. This paper presents a report on the performance of an existing diesel engine for different shims configuration with ethanol and its blends with diesel. As the shim thickness reduction leads to injection advance which is the case required for the use of Ethanol as fuel in C.I engines, the purpose of this paper is to find the best shims configuration for Pure Diesel, E20 and E40 blend for an existing diesel engine. The experiments were conducted on a Direct Injection, 553 cc diesel engine of 5 HP. Experimental tests were carried out to study the configuration on the basis of five performance parameters (M_f, AFR, BSFC, Volumetric and Thermal efficiency). The test results show that the 4 shims (0.66 mm) configuration and 6 shims (1.535 mm) configuration are most suitable for pure diesel and no shims (0 mm) configuration for E20 blend. And increase in performance can be further improved by use of lubricates and igniters along with the change in coolant circulation rate to maintain the optimum working temperature of engine. It is also found that CI engine with less compression ratio are not compatible to run on higher blends of ethanol (E60 and above).

KEYWORDS: Diesel Engine, Ethanol Blends, Fuel Properties, Shims and Performance Parameters

Original Article

INTRODUCTION

Diesel engines have been widely used in almost all walks of life as engineering machinery, automobile and shipping power equipment due to their excellent drivability and economy. At the same time, diesel engines are major contributors of various types of air pollutants such as carbon monoxide (CO), oxides of nitrogen (NOx), particulate matter (PM), and other harmful compounds. With the increasing concern of the environment and more stringent government regulation on exhaust emissions, the reduction in engine emissions is a major research objective in engine development.

Based on the depletion of fossil fuels and environmental considerations has led to investigations on clean and renewable fuels such as ethanol, hydrogen, and biodiesel. Ethanol is regarded as a kind of renewable fuel because it can be made from any kind of ligno-cellulosic materials such as corn, maize, sugar beets, sugar cane, cassava, etc.

The physical and chemical characteristics of ethanol-diesel blends are very important to its application on diesel engine. The stability, density, viscosity, surface tension, specific heat, heat value, and cetane number of blends have great impact on the injection, atomization, ignition, and combustion properties, as well as cold start, power, fuel consumption, and emission characteristics of engine. Additionally, the poking and leakage of

conventional tank, fuel pipe, and sealing part can be rendered. More stringent demands are necessary for the mixture, transportation, storage, and usage of fuel because of low flash point of ethanol-diesel blends [9–13].

Table 1

Property	Diesel	Ethanol
Density	832 kg/m3	789 kg/m3
Auto-ignition temp	256 ^{0}C	363 ^{0}C
Cetane number	40-45	8-12
Viscosity	3.97 CS @ 100^{0}F	1.20 CS @ 100^{0}F

The cetane number is an important fuel property for CI engines. It has an influence on engine start ability, emissions, peak cylinder pressure, and knocking. Alcohol ignites under compression at temperatures approximately 60% greater than diesel fuel does. According to research carried out by Li et al. [12], each 10-vol% ethanol added to the diesel fuel, results in a 7.1-unit reduction in cetane number of the resulting blend. References [8, 14, 15] indicated that the addition of ethanol resulted in increased ignition delay, reduced combustion duration, high maximum pressure rates, and slightly decreased gas temperature because of its low cetane number and low heat value. With the addition of cetane number improver, the combustion properties can reach the level standard engine fuel at medium-high load.

Without modification, the ethanol-diesel blends decreased the power of diesel engine and increased the brake-specific fuel consumption; however, the performance of engine can be rehabilitated after adjusting the fuel delivery and injection timing of engine [16–18]. Reference [19] showed no significant power reduction in the engine operation on different blends of ethanol-diesel (up to 20%). Brake-specific fuel consumption increased by up to 9% as compared to diesel alone. The exhaust gas temperature and lubricating oil temperatures were lower with operations on ethanol-diesel blends as compared to operation on diesel alone.

Ethanol-diesel blends can reduce the smoke and PM emissions of diesel engine. The higher the percentage of ethanol is in the blends, higher is the reduction in smoke and PM emissions. The reason is that the oxygen content in blends can promote the combination of fuel and oxygen, even in fuel-rich region [16, 20–22]. The NOx emissions remained the same or very slightly reduced with the use of the ethanol-diesel fuel blends with respect to those of the diesel; however, the NOx emissions can be reduced by other techniques, such as EGR and SCR. The hydrocarbons (HCs) emissions were increased with the use of ethanol-diesel blends. The higher the percentage of ethanol in the blend, higher is the increase in HC emission, however, the HC emissions of blends can still meet the emission standards due to low HC emissions of diesel engine. References [12, 20] showed that the CO emissions of ethanol-diesel blends were increased at low load and were decreased at high load. Additionally, the CO_2 emissions were decreased due to the low C/H ratio of ethanol-diesel blends.

Back in the fuel-crisis days of the late 1970s and early 1980s, so-called fumigation systems were popular in agricultural circles. Those involved after-market adaptations; used carburettors mounted on the engine's intake manifold. After adjusting the injection system to cut back fuel delivery and tying in the carburettor's throttle valve to match the tractor's load, the engine essentially took between 30 and 50 percent of its fuel charge from the carburettor, depending on load. At idle, the engine consumed diesel almost exclusively. For turbocharged diesels, an alcohol injection system was developed at around the same time by an outfit called M&W Gear Company. This used an alcohol injector mounted downstream from the turbocharger. It discerned turbocharger pressure and delivered the appropriate amount of fuel to lower diesel consumption and boost horsepower. Early in 2008, the Swedish automotive-tech company BSR Svenska AB

unveiled their ethanol-powered diesel conversion, a Saab 9-3 optimized for E-95 (95 percent ethanol, 5 percent gasoline) fuel. The engine's combustion chamber was modified and the fuel system and ECU software altered to suit the properties of alcohol. The 195-horsepower engine achieved fuel mileage figures of around 46 miles per gallon and reduced carbon dioxide and hydrocarbon emissions by approximately 95 percent. This automobile is not in production, but has provided a clear path for future development. The company markets E-85 conversion kits for a variety of gasoline cars.

From the above literatures it is clear that ethanol has been used as a fuel for compression ignition (CI) engines since 19th century but the easy availability of gasoline has led us to believe that the petroleum-derived fuel is the only one suitable for internal combustion engine. For decades, the petroleum industry has dedicated itself to promote its products and has built a massive transportation and industrial infrastructure which is taking so long to the Govt lobby to switch to renewable fuel sources. Utilization of ethanol in diesel engine can be operated such by blending and fumigating process. The optimal conditions for engine performance depend strongly on the specific configuration. Therefore, technological advances and research activities in this field must go on, particularly for the fact that diesel emissions can be significantly reduced with the addition of alcohol in combustion. So, the objective of this study is to understand ethanol- diesel blend performance in CI engines by compiling different performance parameters at various engine loads and ethanol percentages with four different injection timings. And the injection timing was achieved by changing the compression ratio of fuel injection pump. Four different shims configuration were used for the experimentation. The effects were studied by different plots and computation of % change of performance parameters like fuel consumption, brake specific fuel consumption and brake thermal efficiency verses power output.

Experimental Setup and Procedure

This study was conducted on a single cylinder four stroke direct injection diesel engine with a mechanical in-line fuel pump of Kirloskar make. Single cylinder 5 HP engine was chosen as this was light and easy to maintain. Being water cooled, the system is most suitable for hot and arid conditions. The objective of this paper was to replace diesel fuel to the maximum without much loss of performance. The standard compression ratio of the engine was 16.5:1. It was fitted with a Hartnell governor to maintain a rated speed of 1500 rpm. The crankshaft of the engine was connected to a drum with rope brake arrangement to measure the power output. Readings were taken with no load, 2 kg, 4 kg, 6 kg and 8 kg loading conditions. Digital tachometer was used to measure the speed at specific loading condition. Calibrated burette and a stop watch was used to measure the volume flow rate of fuel. AFR was measured using air box with orifice diameter of 24mm and had a volume of 100 times the swept volume of the engine. U-tube manometer of 50 ml was used to measure pressure difference. RTD's were used to measure inlet and exhaust air temperatures and also the inlet and outlet water temperatures, whose values were used to calculate the heat balance sheet.

Figure 1: Experimental Setup
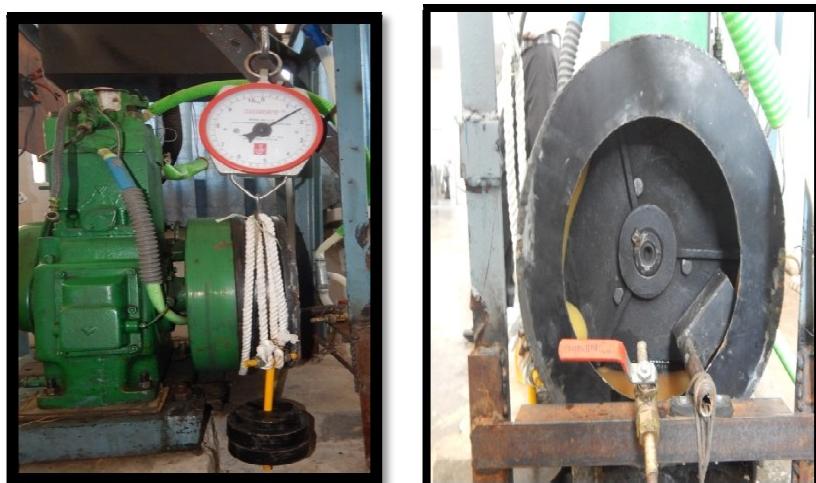
Rope Brake Dynamometer   **Drum Cooling Setup**

Figure 2

Table 2

Engine Specifications	
Model	**Rocket Engg Model VRC1**
Bore / Stroke	80 mm / 110 mm
Swept Volume	553 cc
Rated R.P.M.	1500
B.H.P.	5 H.P.
Compression Ratio	16.5:1
No. of cylinders	1
Type of fuel injection	Direct injection

The fuels used in the study are standard diesel (density = 832.7 kg/m^3) and ethanol (density = 789 kg/m^3). The blending was done on volume basis with three blend ratios of 20%, 40% and 60% of ethanol with 80%, 60% and 40% diesel respectively. Commercial diesel fuel and anhydrous ethanol was used for preparation of the blends. The experiments were conducted under steady state for five different loading conditions (No load, 2kg, 4kg, 6kg and 8kg) each for three different proportions of blend. All data were collected after the engine was stabilized after running for 5minutes. This steady state tests were repeated four times for each set of condition to ensure like results. Results have been drawn on the average values and analyzing them in terms of % change w.r.t. pure diesel.

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Table 3

Fuel\Property	Calorific Value (KJ/Kg)	Density (Kg/m^3)
Pure diesel (E0)	44800	832.7
20% ethanol + 80% diesel (E20)	41780	823.96
40% ethanol + 60% diesel (E40)	38760	815.22
60% ethanol + 40% diesel (E60)	35740	806.48
80% ethanol + 20% diesel (E80)	32720	795.94
Ethanol (E100)	29700	789

Modification

Cetane number of ethanol 8-12 is very low as compared to that of diesel which is 40-45. And has a high self burning temperature compared to diesel. This prevents their ignition at the end of compression stoke because of which ignition delay is more in case of ethanol which was calculated using the equation below:

$$\overline{\tau}id(CA) = (0.36 + 0.22Sp)\exp[EA(\frac{1}{RT} - \frac{1}{17190})(\frac{21.2}{P-12.4})^{0.63}]$$

Where,

P - pressure at the end of compression stroke at TDC (bars)

T - temperature of the charge at TDC (Kelvin)

Sp - mean piston speed of the engine (meter per second) = 5.5 m/sec

R - universal gas constant (8.314 J/mol K)

EA - apparent activation energy (Joules per mole) and is given by

$$E_A = \frac{618840}{CN + 25}$$

CN - Cetane number of the fuel

Ignition delay for diesel and ethanol is found to be 3.72 ms and 7.82 ms respectively. This difference of 4.1 ms is because of their cetane number. Pure ethanol takes maximum time to burn but the blends E20 and E40 will require lesser delay, thus performance of engine is evaluated at various injection timing. In conventional inline pump which is varied using different shim thickness. It increases compression ratio of fuel pump and injection pressure is achieved at different advance timing. Different injection timings were achieved by manipulating the plunger lift in the fuel injection pump by varying shim thickness. Four different shims configuration were used for the experimentation and the effects were studied by different plots and computation of % change of performance parameters like fuel consumption, brake specific fuel consumption and brake thermal efficiency verses power output.

SHIM COMBINATION USED	
No. of Shims	THICKNESS (MM)
0	0
2	0.33
4	0.66
6	1.535
10	2.195

Figure 3

RESULT AND DISCUSSIONS

Results which are engine performance based on parameters such as brake thermal efficiency, brake specific fuel consumption, fuel consumption and exhaust gas temperature were recorded for different shim configuration at static loading condition. Results calculated from those parameters are plotted on graph to do comparative study of performance.

Pure Diesel
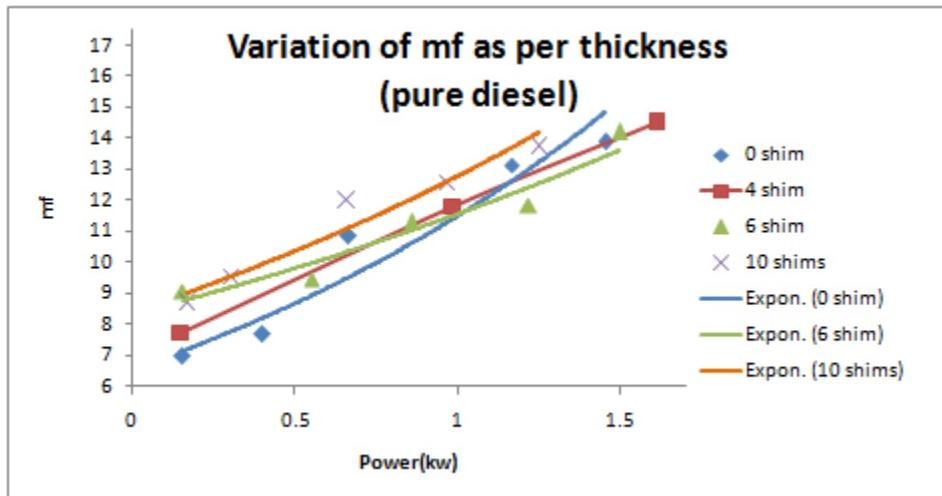
Figure 4

The trends for consumption of fuel with respect to power make it a bit difficult to conclude any specific shim configuration for pure diesel. The lines are intersecting each other so the area under the curve was found to have a better prospect of the trend and thus the fuel consumption is found out to be minimum for 4 and 6 shim configuration equally.
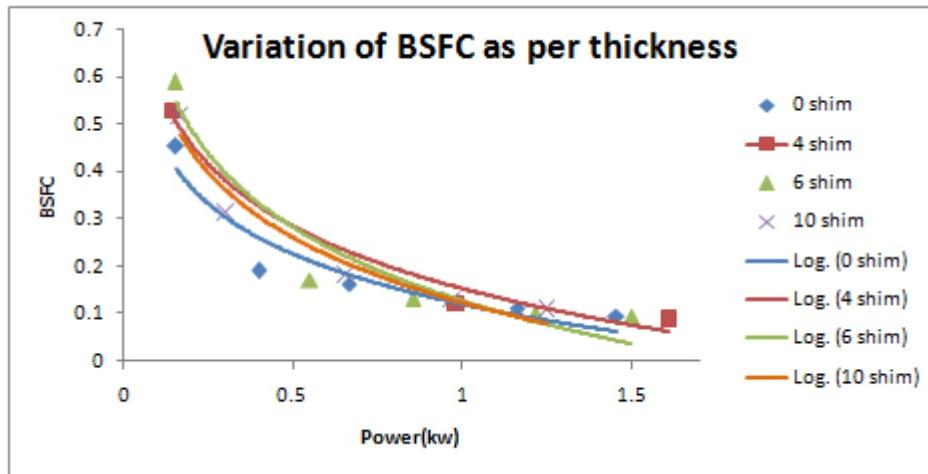
Figure 5

Ideally BSFC should be decreasing as the loading increases and for different configurations here the brake specific fuel consumption shows similar trend with pure diesel. But the area under BSFC curve is found minimum for 0 shim configuration and moderate in case of 4 and 6 shims configurations; 6 being comparatively lower.
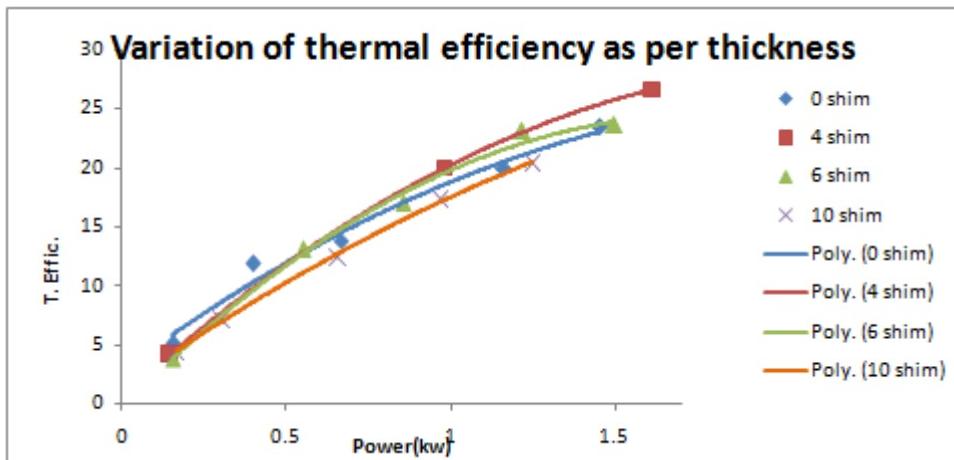

Figure 6

Especially under high loading condition the injection timing is best suited to burn the fuel completely and deliver peak piston pressure at right time. Although the fuel flow rate in 0 shim configuration is least as the duration of injection is less but has comparatively low brake thermal efficiency, so for pure diesel 4 shim configuration is best suited.

E20
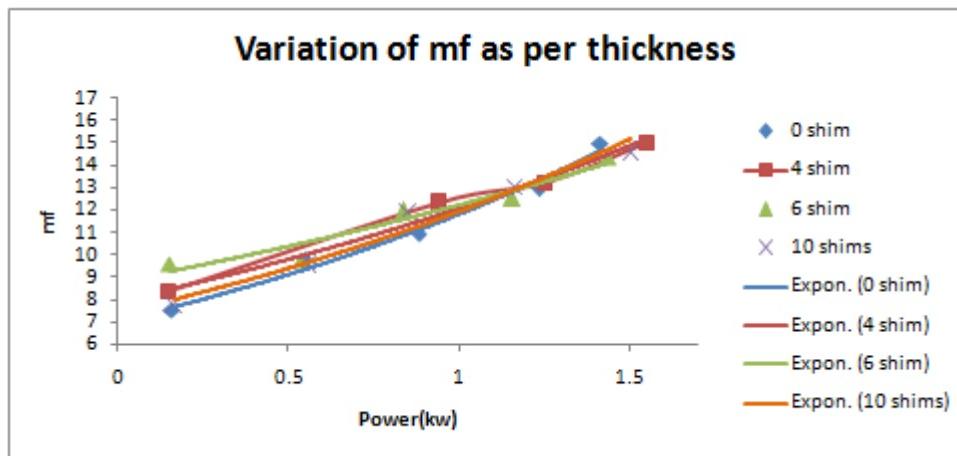
Figure 7

Being a considerable difference in calorific value, the amount of fuel needed to generate same power as pure diesel; the quantity of blends needed will be more. So, for E20 (20% ethanol and 80% diesel) m_f for the standard configuration for pure diesel must be higher, which is clearly justified with the area under the curve for each shim configuration for both blends at specific power range. And for E20, 0 shim configuration proved to be the best as fuel consumption was minimal here.
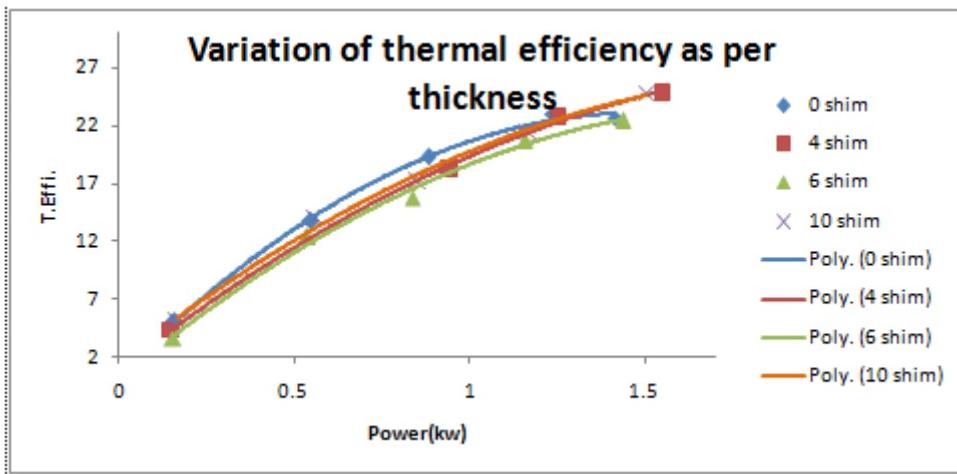

Figure 8

There is a shift in combustion process i.e. ignition delay decreased for various shim configuration from low load to high load. As the load increases, fuel supply increases which mean for any injection advance, more fuel will be supplied before the combustion starts and which initiates combustion before ideal crank position and apparently efficiency goes down with increasing load. This concept can be seen in the trend with 0 and 10 shim configurations, 0 shim having higher efficiency. In case of 10 shim configuration the advance is less thus the initiation of combustion initially results in peak pressure generation at ideal crank position.

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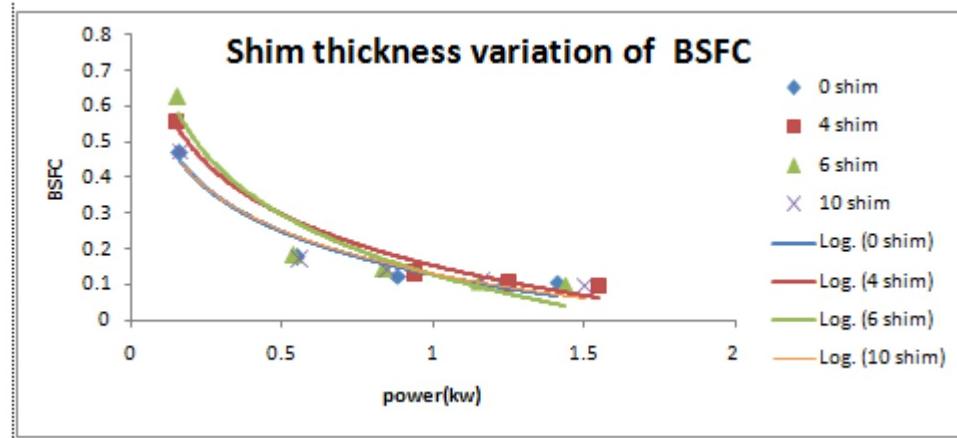
Figure 9

Construing the trend of 0 shim configuration for brake thermal efficiency and fuel consumption, BSFC is expected to be low for 0 shim configuration but it increased comparatively to other configurations with the increase in load. Under high loading condition BSFC is minimal for 6 shims configuration. So, it can be easily lined up that ignition delay varies with loading condition and in order to use E20 we need more advance than pure diesel. When area under this curve is calculated, it was found that 0 shim configuration suits best for E20 over other advance positions.

E40
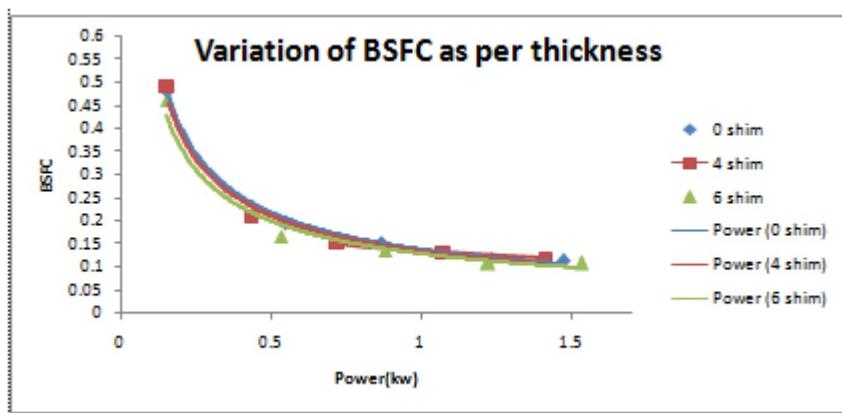
Figure 10

For E40, the overall fuel consumption must be high comparing with pure diesel and E20 blend. Here BSFC has also shown slight increment at low load condition but at higher load the trend for all shim configurations are more or less same i.e., there is no effect of loading on BSFC. It must also be noted that the test engine didn't run with 10 shims configuration for E40.
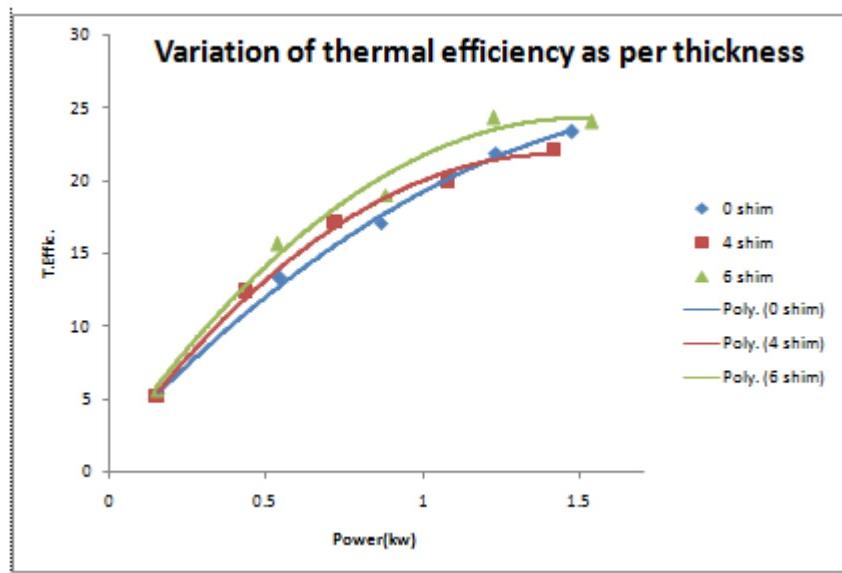
Figure 11

This graph clearly shows that for E40, 6 shims configuration has maximum brake thermal efficiency right from no load to full load condition. The maximum efficiency achieved at any shim configuration has comparatively dropped by 2-3% compared to previous results for pure diesel and E20.

CONCLUSIONS

- It was observed that existing CI engine which has been designed to run on pure diesel can run on E20 blend with little advance in injection timing. Injection timing advancement is possible with all diesel engines having their own separate means for change.

- M_f and BSFC were found optimal for pure diesel (E0) at 6 shims configuration, whereas for E20 minimum 0 shim configuration was observed best.

- For particular injection timing, brake thermal efficiency came out to be different for different fuel. So it is better to have variable injection timing or the injection timing must be set to give maximum efficiency under high loading. It reduces the fuel consumption considerably and with this intent 6 shims configuration suits best for pure diesel and 0 shim configuration for E20.

- Minute Knocking is observed while operating on E40 blend which was because the self ignition temperature for the blend is considerably higher than pure diesel and E20. Moreover, efficiency comparatively came down in case of E40 and overall fuel consumption has increased because of its lower calorific value and inefficient running.

- The test engine when tried with E60 showed violent vibrations and continuous knocking and it stopped working on various injection advances. It was because the pressure in combustion chamber was not enough to raise the temperature to self ignition temperature of the blend and a lot of fuel was getting collected in highly compressed air before combustion would start which was resulting in uncontrolled vibrations and knock. Possible Solution to this problem is to increase the compression ratio so that the self ignition temperature is attained at appropriate time. Or a spark plug along with the injector can be added to initiate combustion at desired time with same compression ratio.

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